

April 6, 1954

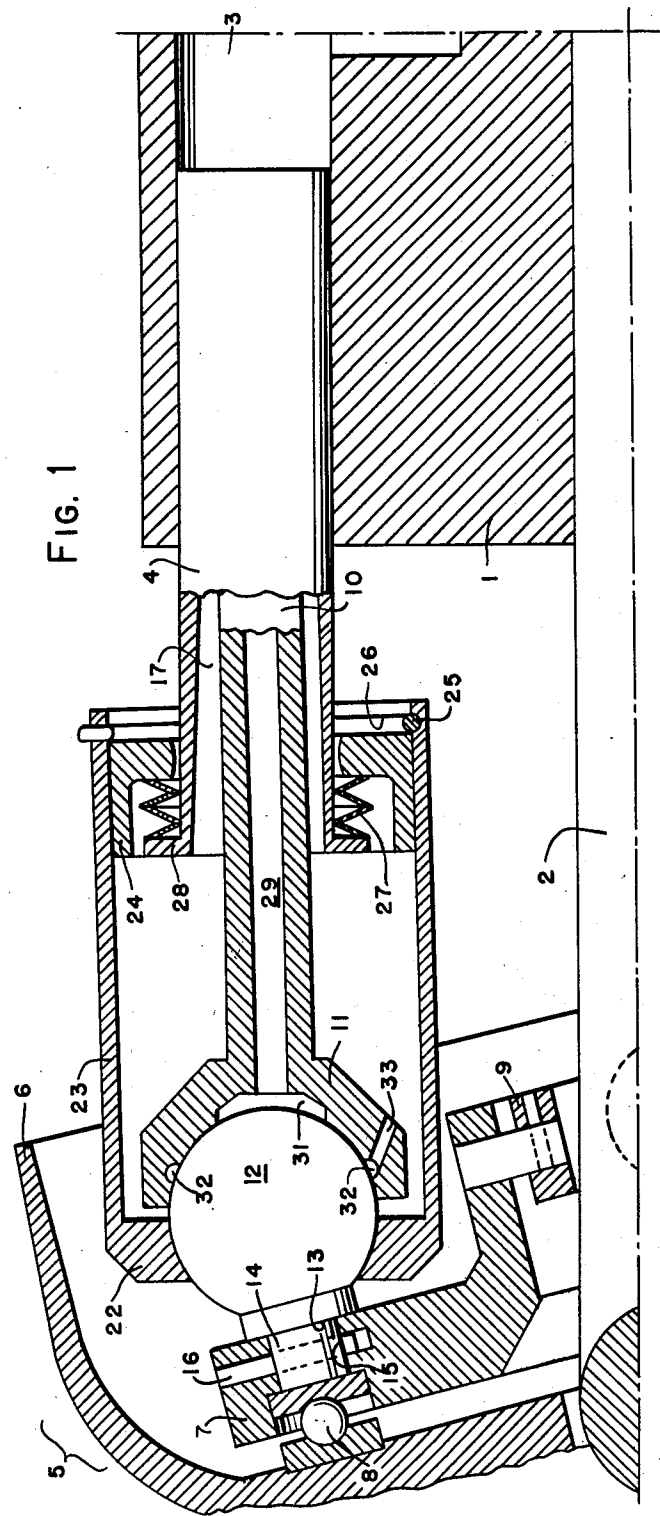
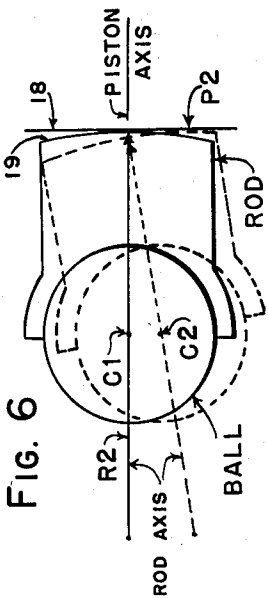
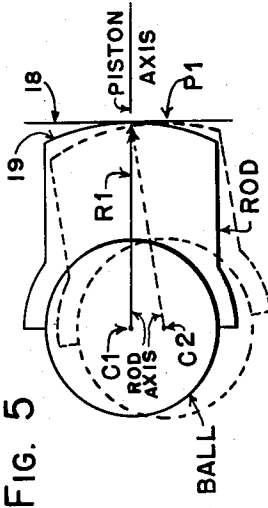
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2,674,196

PISTON ASSEMBLY FOR AXIAL TYPE HYDRODYNAMIC MACHINES

Filed Oct. 10, 1951

2 Sheets-Sheet 1



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2 Sheets-Sheet 2

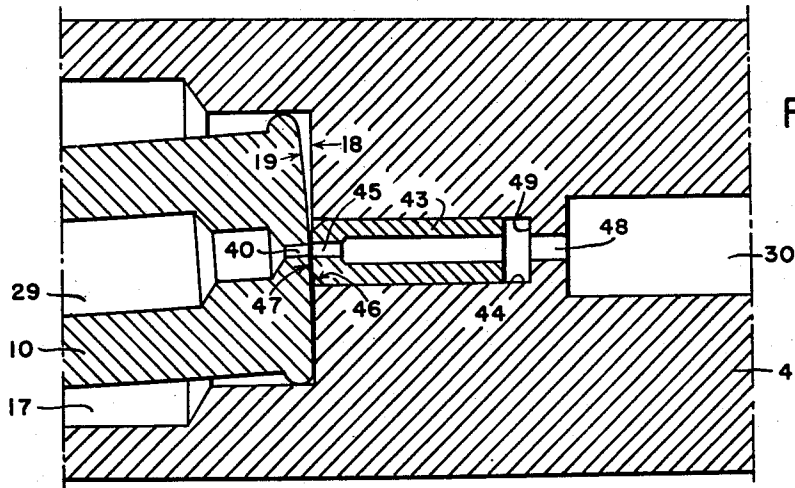


FIG. 2

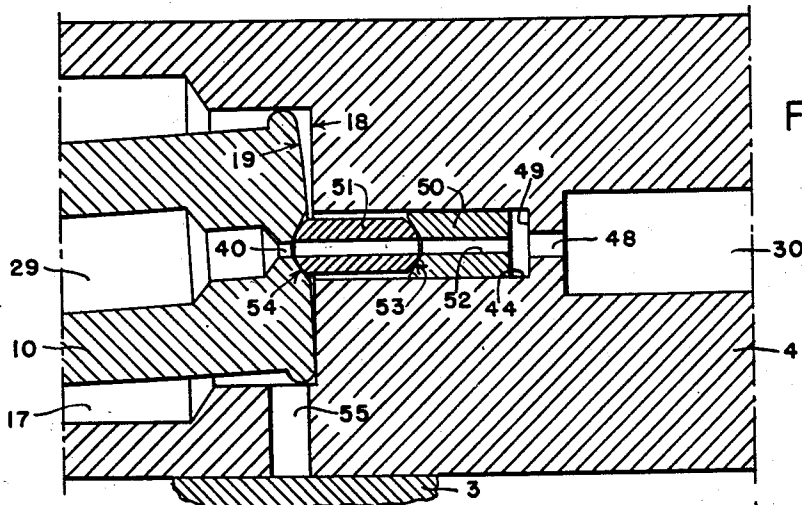


FIG. 3

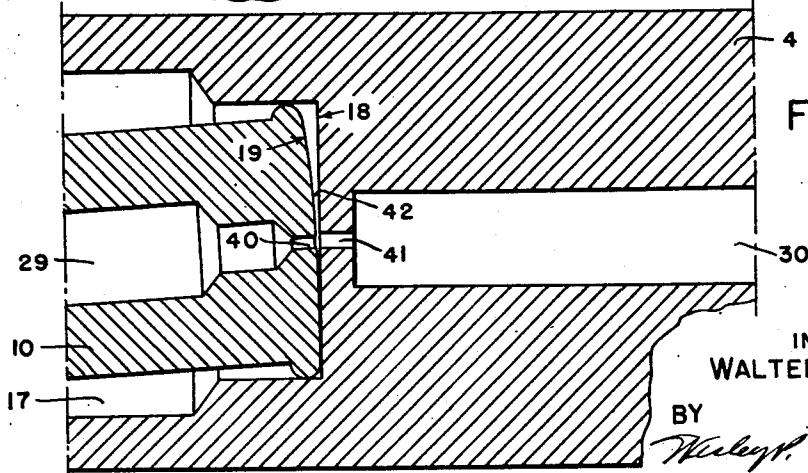


FIG. 4

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PISTON ASSEMBLY FOR AXIAL TYPE
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7 Claims. (Cl. 103—162)

1

This invention relates to piston assemblies of the type shown in Patent No. 2,638,850 and in abandoned application Serial No. 711,674, filed November 22, 1946 of which this application is a continuation-in-part.

Such piston assemblies constitute parts of a hydrodynamic machine of the type in which a plurality of cylinders are arranged in a cylinder barrel, a piston is fitted in each cylinder and each piston is connected to a thrust member by a piston rod. A machine of that type will function as a pump when driven mechanically and will function as a motor when supplied with liquid under pressure. In order to simplify the following description, the invention will be explained as being incorporated in a pump but, since the same machine will function either as a pump or as a motor, it is to be understood that the invention is equally applicable to motors and that the appended claims are in no way limited to piston assemblies for pumps.

Piston assemblies embodying the invention may be incorporated in a number of types of pumps. For example, the pump may be of the well known swash plate type in which the cylinder barrel is rotated by a coaxial drive shaft and the thrust member or swash plate is inclined to and driven by the drive shaft, it may be of the well known angle type in which the thrust member is normal to and driven by the drive shaft and the cylinder barrel is inclined to and driven from the drive shaft, or it may be as a well-known wobble plate type in which the cylinder barrel is stationary and the thrust member or wobble plate is oscillated or wobbled by a member which is inclined to the cylinder barrel axis and is rotated by the drive shaft. It is only necessary that the pump be of such a type that any given point on the face of the thrust member moves toward and from the cylinder barrel when the pump is operated.

In swash plate, angle and wobble plate types of pumps, the pistons and cylinders are always arranged in a circle with the axes of the cylinders parallel to each other but, due to the plane of the thrust member being inclined to the cylinder axes when the pump is pumping liquid, the extended centerlines of the pistons intersect the plane of the thrust member in an ellipse instead of a circle. It has been common practice to provide piston rods between the several pistons and the thrust member. The outer ends of such rods are connected to the thrust member upon a circle having a radius nearly the same as the radius of the circle in which

2

the cylinders are arranged. The outer end of each piston rod has a gyrating motion about the piston axis as the pump operates, thus compensating for the difference between the radius of the cylinder circle and the axes of the above described ellipse on the plane of the thrust member.

In the prior pumps, each piston rod ordinarily has its outer end connected to the thrust member by a ball and socket joint and its inner end connected to the piston by another ball and socket joint which is arranged within the piston far enough from the outer end thereof to cause the pumping forces transmitted from the thrust member through the rod to be applied to the piston at a point which is intermediate the ends of the cylinder in all positions of the piston.

As the outer end of a piston rod gyrates, the ball on the inner end of the rod rotates in its socket through a limited angular distance and, in order to prevent the inner ball and/or its socket from being abraded by the relative movements thereof, means are provided for lubricating the joint. This is ordinarily accomplished by drilling a small hole from the inner end of the piston through the socket so that the pressure created in the cylinder forces liquid to flow through the hole and spread laterally between the mating surfaces, thus providing a lubricating film between the inner ball and its socket.

When the pump is in operation, the thrust member forces each piston inward during one-half of each revolution of the cylinder barrel and draws the piston outward during the other half of each revolution. Each inward moving piston ejects liquid from the cylinder and, when the ejected liquid meets a resistance, pressure is created in the cylinder. The force required to create this pressure is equal to the cross sectional area of the piston multiplied by the unit pressure and, since this force must be transmitted through the inner ball and socket joint, it must not exceed the bearing value of the lubricating film in that joint as otherwise the film would break down and abrasion of the joint would occur.

The bearing area of the inner ball is necessarily only a fraction of the piston area for the reason that the ball is inside the piston. Therefore, the pressure created by the piston must not exceed a given low value for the reason that the force necessary to create a high pressure over the relatively large piston area would break down the lubricating film in the relatively small area between the ball and its socket and would

thus cause abrasion of the ball and the socket joint.

In each of the piston assemblies shown in the above application, the piston has an axial bore extending into it from its outer end and a flat bearing surface at the inner end of the bore, a piston rod extends into the bore and has a spherical face on its inner end to engage the bearing surface on the piston, the outer end of the piston rod is connected to a thrust member by a ball and socket joint which is larger in diameter than the piston, a recess of substantial area is formed in one of the mating surfaces of the ball and socket joint on the axis of the piston rod, and various means are provided for supplying liquid to the recess from the liquid in the cylinder. When the pump is in operation, liquid seeps from the recess and forms a lubricating film between the surface of the ball and the socket which are in contact with each other.

A large part of the force necessary to create a given pressure in the cylinder is transmitted from the thrust member to the connecting rod through the liquid in the recess and the remainder of that force is transmitted through the portion of the lubricating film surrounding the recess. Since the pressure in the recess is substantially the same as the pressure in the cylinder and since the ball and socket joint is larger in diameter than the piston, a very large force can be transmitted from the thrust member to the piston rod. The end face of the piston rod and the bearing surface on the piston are in rolling contact and do not require lubrication. Consequently, the force transmitted from the rod to the piston may be equal to the bearing value of the metal itself. The result is that the piston assembly is capable of transmitting from the thrust member to the liquid in the cylinder a force which is several times as great as any force which could be safely transmitted through the piston assemblies of the prior pumps.

The present invention has as an object to provide a piston assembly of the above type having improved means for supplying liquid from the cylinder to the ball and socket joints.

The invention is exemplified by the piston assembly shown in the accompanying drawings in which the views are as follows:

Fig. 1 is a vertical longitudinal section through a part of a pump having incorporated therein a piston assembly in which the invention is embodied.

Fig. 2 is a view showing a portion of the piston assembly shown in Fig. 1 drawn to a larger scale to illustrate the details of a sealing member which prevents any substantial amount of liquid from leaking into the bore in the piston the inclination of the piston rod relatively to the piston being greatly exaggerated.

Fig. 3 is a view similar to Fig. 2 but showing a different sealing member.

Fig. 4 is a view showing a portion of a piston assembly which is not provided with any sealing member.

Figs. 5 and 6 are diagrams each of which illustrates two positions of a piston rod relative to the piston with which it is associated, the diagrams being greatly disproportioned.

For the purpose of illustration, an embodiment of the invention has been shown in Fig. 1 as being incorporated in a pump of the swash plate type such as shown in Patent No. 1,044,838.

As shown, the pump has a cylinder barrel 1 fixed for rotation with the drive shaft 2 and

provided with a plurality of cylinders 3 which are equally spaced around shaft 2 and are parallel to the axis thereof, only one cylinder being shown. A piston 4 is fitted in each cylinder and connected to a swash plate 5 which is inclined to the axis of cylinder barrel 1 when the pump is pumping liquid.

Swash plate 5 has been shown as including a tilting box 6 which is pivoted upon trunnions (not shown), an annular thrust member 7 which is rotatably supported in box 6 by a bearing 8, and a universal joint 9 which connects thrust member 7 to shaft 2 so that thrust member 7 and cylinder barrel 1 are rotated in unison by shaft 2. However, if the pump is of large capacity or is to create very high pressures, thrust member 7 is supported in box 6 by one or more mechanical bearings and a hydraulic thrust bearing as shown in Patent No. 2,577,242. Each cylinder 3 is adapted to be connected to opposite sides of an external circuit alternately by a flat faced valve (not shown) which engages the rear end of cylinder barrel 1 according to the usual practice.

When swash plate 5 is inclined to the axis of cylinder barrel 1 and shaft 2 is rotating cylinder barrel 1 and thrust member 7, each piston 4 will be moved outward and will draw liquid into its cylinder 3 during one-half of each revolution of cylinder barrel 1 and it will be forced inward by thrust member 7 and will eject liquid from its cylinder 3 during the other half of each revolution of cylinder barrel 1.

The force for moving each piston 4 into its cylinder is transmitted thereto from thrust member 7 through a piston rod 10 which is connected at its outer end to thrust member 7 by a ball and socket joint which is larger in diameter than piston 4 and the several ball and socket joints are arranged in a circle concentric with the axis of thrust member 7.

The ball and socket joint has been shown as including a socket 11, which is fixed upon the outer end of rod 10, and a ball 12 which is closely fitted in socket 11 and is fixed to thrust member 7 in any suitable manner. As shown, ball 12 is provided with a bearing surface 13 to engage the face of thrust member 7 and with a cylindrical extension 14 which is fitted in a bore 15 in thrust member 7 and is held therein by a pin 16.

Instead of the ball and socket joint having its socket fixed to the piston rod and its ball fixed to the thrust member, it may have its socket fixed to the thrust member and its ball fixed to the piston rod as shown in Patent No. 2,638,850. Also, the ball and socket joint may be fixed to the thrust member of a different type of pump such as the angle type pump or the wobble plate type pump.

Piston rod 10 extends into a bore 17 which is formed in piston 4 and which is enough larger in diameter than rod 10 to permit rod 10 to tilt relatively to the piston axis such as by making bore 17 larger at its outer end than at its inner end, the taper of the bore being exaggerated in the drawing.

The inner end portion of the rod may be larger in diameter than the intermediate portion of the rod and it is closely fitted in the inner end portion of bore 17 with sufficient clearance to permit rod 10 to tilt relatively to the piston axis. As shown, the inner end wall of bore 17 is normal to the axis of piston 4 and provides a substantially flat bearing surface 18 (Fig. 2) which is engaged by a spherical end face 19 formed upon the inner end of rod 10. Instead of bearing sur-

face 18 being flat and end face 19 being spherical, end face 19 may be flat and surface 18 may be spherical or both the end face and the bearing surface may be slightly spherical as it is only necessary that end face 19 have a rolling contact with surface 18. End face 19 engages bearing surface 18 upon a single contact spot which moves toward, from and around the piston axis when cylinder barrel 1 and thrust member 7 are rotated and thrust member 7 is inclined to the axis of cylinder barrel 1.

If the piston assembly is incorporated in a machine in which the pistons are moved outward hydraulically such as a machine which functions as a motor or a machine which is employed as a pump in a closed non-differential circuit and is supercharged with liquid from a small auxiliary pump in the well known manner, it is not necessary to provide mechanical means for retracting the pistons but the piston assembly should include means, such as shown in Patent No. 2,638,850, for keeping socket 11 and ball 12 in contact with each other so that the ball and the socket cannot separate during assembly or when there is no pressure in the cylinder.

If the piston assembly is incorporated in a pump which must suck liquid into its cylinders, it is necessary to provide mechanical means for retracting the pistons. In order that piston 4 may be pulled outward on a suction stroke, an annular spherical socket 22 is fitted upon ball 12 and provided with a cylindrical extension 23 which encircles socket 11 and the outer end portion of piston 4. An annular member 24 is fitted in extension 23 around piston 4 and retained in position by a snap ring 25 which is fitted in a groove 26 formed in the inner peripheral surface of extension 23. Annular member 24 is engaged by suitable spring means 27, such as a stack of Belleville washers, arranged between it and a peripheral flange 28 formed upon the end of piston 4.

Annular member 24 is bored to slip freely over piston 4 and the inner wall of its bore is rounded to permit extension 23 to tilt relatively to piston 4 as ball 12 moves relatively to the piston axis during operation of the pump. Spring means 27 has sufficient strength to hold socket 11 against ball 12 and surface 18 against end face 19 while permitting face 19 to rock or roll upon surface 18 as ball 12 moves relatively to the piston axis during operation of the pump.

Liquid for lubricating the ball and socket joint is supplied thereto through a passage 29, which extends axially through rod 10, and a passage 30 (Fig. 2) which extends from bore 17 through the inner end of piston 4. Passage 29 communicates with a recess 31 which is concentric with the axis of rod 10 and has a diameter less than the diameter of piston 4. Recess 31 has been shown as being formed in the spherical surface of socket 11 but it would be formed in ball 12 if ball 12 were on rod 10.

When piston 4 is creating pressure in cylinder 3, liquid will flow from cylinder 3 through passages 30 and 29 to socket 11 and form a lubricating film between it and ball 12. The working pressure will extend into recess 31 and into the lubricating film but there will be no pressure in recess 31 or in the film when there is no pressure in the cylinder.

The liquid in this lubricating film and in recess 31 when under pressure will exert a "blow-off force" which tends to move socket 11 away from ball 12. In order that this blow-off force may

not become so great that it will force the mating surfaces of ball 12 and socket 11 apart and thus permit substantial leakage losses of the working liquid, an annular pressure limit groove 32 having a diameter greater than piston 4 is cut around socket 11 in a plane normal to the axis of rod 10 and a drain duct 33 is extended from groove 32 outward through socket 11 to permit liquid to escape freely from groove 32.

Pressure is thus limited to the area between recess 31 and groove 32 and the film in this area may be designated as a "pressure lubricating film" but there is no pressure in the lubricating film between ball 12 and the outer part of socket 11 nor in the lubricating film between ball 12 and socket 22. The area of recess 31 and the area of the pressure lubricating film should be such that the total blow-off force is less than the force exerted by piston 4 upon the liquid in cylinder 3.

End face 19 on rod 10 preferably has a very long radius in order that the spot thereon which contacts bearing surface 18 may have a very high bearing value and in order that the contact spot may be moved a substantial distance from the axis of piston 4 when rod 10 is tilted relatively to the piston axis.

It will be obvious that the contact spot would be on the axes of the piston and the rod if those axes were coincident with each other and, if the axis of the rod is tilted out of alignment with the piston axis, the center of the contact spot will shift away from the piston axis a distance which is proportional to the radius of surface 19 and the angle between the rod and the piston axes as illustrated diagrammatically in Figs. 5 and 6 in which the length of a piston rod relatively to the diameter of the rod has been reduced to a small fraction of the actual length and the indicated movement of the center of the ball and socket joint has been exaggerated in order to prevent certain lines from merging with each other.

For example and as indicated in Fig. 5, if end face 19 had a radius R_1 struck from the center of the ball and socket joint and the center of the ball and socket joint were shifted from the position C_1 on the piston axis a given distance to position C_2 , the center of the contact spot would shift from the piston axis to a point P_1 which is quite close to the piston axis so that the contact spot would move around over a relatively small area. But if the center of the ball and socket joint were shifted the same distance from position C_1 to position C_2 and if end face 19 had a radius R_2 twice the length of radius R_1 as indicated in Fig. 6, the center of the contact spot would shift to a point P_2 which is twice as far from the piston axis as is point P_1 so that the contact spot would move over a larger area.

Preferably, the construction is such that the axis of the rod is never coincident with the axis of the piston so that the contact spot will not pass onto the axis of the piston nor onto the axis of the rod because lubrication passages 29 and 30 are on those axes.

If the centers of the ball and socket joints are arranged in a circle having a radius which is smaller than the radius of the circle upon which cylinders 3 are arranged or which is greater than the major axis of the ellipse defined by the extended axes of the cylinders on the plane of the ball and socket joint centers, the axes of each piston and cylinder are never coincident with each other and, consequently, the center of the contact spot is never on the piston axis. If the centers of the ball and socket joints are arranged

on a circle having a radius which is greater than the radius of the circle upon which cylinders 3 are arranged and is smaller than the major axis of the ellipse mentioned above, the axes of each piston and cylinder are not coincident with each other except when swash plate 5 is adjusted to one particular position intermediate its minimum and maximum displacement positions and the piston is at the end of its stroke.

However, it has been found from experience that, if the portions of passages 25 and 30 adjacent to end face 19 and to bearing surface 18 are made as small in diameter as is practical, no appreciable deformation of surfaces 18 and 19 will be caused by the contact spot passing rapidly back and forth across the piston and rod axes for a very long period of time when the forces transmitted through the contact spot are great enough to enable the piston to create a pressure of 5000 p. s. i. in its cylinder.

The piston rod assembly thus far described is substantially the same as one of the piston rod assemblies shown in application Serial No. 711,674 to which reference may be had for further details and explanation.

The present invention provides novel and inexpensive means for preventing any substantial amount of liquid from leaking out of passages 29 and 30 into bore 17. The end portion 40 of passage 29 is made quite small as the flow of liquid therethrough is very limited but it must be large enough to prevent it from becoming plugged with foreign matter which often accumulates in the motive liquid. If passage 30 had an end portion 41 of substantially the same diameter as portion 40 as shown in Fig. 4, there would be an opening 42 between surface 18 and end face 19 around the ends of passage portions 40 and 41 when rod 10 was inclined to the axis of piston 4 and liquid could escape through opening 42 into bore 17 at too great a rate. But the escape of liquid into bore 17 may be greatly reduced or eliminated by providing in passage 30 a sealing member which is urged against the end of rod 10 by the pressure in cylinder 3.

For example and as shown in Fig. 2, a sealing member 43 may be closely fitted in the end portion 44 of passage 30 and provided with a small axial passage 45 in alinement with passage portion 40 so that liquid may flow from cylinder 3 through piston 4 and rod 10 to the ball and socket joint. Any pressure in cylinder 3 acts upon the rear end of sealing member 43 and urges the front end thereof against end face 19 on rod 10. Preferably, sealing member 43 is made as small in diameter as is practical and the front end thereof is chamfered, as indicated at 46, to reduce the end face of member 43 to a very narrow annular contact surface 47 around the end of passage 45.

When the axis of rod 10 is inclined to the axis of piston 4, contact surface 47 will engage end face 19 at one side only of passage 45 but contact surface 47 is so small in diameter that the clearance between contact surface 47 and end face 19 at the other side of passage 45 is so little that the amount of liquid which can escape therethrough into bore 17 is so small as to be negligible even when rod 10 is at its maximum inclination in respect to piston 4.

In order that sealing member 43 may not move too far away from its normal position during assembly of the pump or when there is no pressure in cylinder 3, a portion 48 of passage 30 at the rear end of portion 44 is reduced to a diam-

eter less than the diameter of portion 44 to provide a shoulder 49 which limits the distance sealing member 43 can move away from its normal position.

Fig. 3 shows the piston assembly provided with a sealing member composed of a first part 50, which is closely fitted in portion 44 of passage 30, and a second part 51 which is loosely fitted in passage portion 44. A small axial passage 52 extends through both parts of the sealing member in alinement with passage portion 40 so that liquid can flow from cylinder 3 through piston 4 and rod 10 to the ball and socket joint. The rear end of part 51 is made spherical and closely fitted in a conical recessed seat 53 formed in the front end of part 50 and the front end of part 51 is shaped to fit into a conical recessed seat 54 formed in the end of rod 10 around passage portion 40. The front end of part 51 has been shown as being spherical but it may be chamfered as shown at 46 in Fig. 2.

The arrangement is such that any pressure in cylinder 3 will urge part 50 against part 51 and part 51 against rod 10. Part 50 forms a substantially fluid tight seal with the wall of passage portion 44, the spherical rear end of part 51 forms a substantially fluid tight seal with seat 53 and if the front end of part 51 is spherical, it will form a substantially fluid tight seal with seat 54 so that substantially no liquid can escape into bore 17.

While no lubrication of bearing surface 18 and end face 19 is needed, the outside of rod 10 and the walls of bore 17 should be coated with liquid to prevent rusting and to prevent abrasion of the periphery of the inner end of rod 10 and the inner portion of the peripheral wall of bore 17. In the arrangement shown in Fig. 2, sufficient liquid for this purpose will escape into bore 17 from between sealing member 43 and end face 19. Since in the arrangement shown in Fig. 3 substantially no liquid can escape past the sealing member into bore 17, piston 4 is provided at the inner end of bore 17 with a small radial hole 55 through which liquid can flow into bore 17 at a minute rate from the film of liquid between piston 4 and the wall of cylinder 3.

The invention herein set forth may be modified in various ways without departing from the scope thereof as defined by the following claims.

I claim:

1. A piston assembly for a hydrodynamic machine having a cylinder and means including a thrust member for reciprocating a piston in said cylinder, said piston assembly comprising a piston fitted in said cylinder and having an axial bore extending into it from its outer end, a bearing surface on said piston at the inner end of said bore, a piston rod extending into said bore and having upon the inner end thereof a contact surface which engages said bearing surface, one of said surfaces being spherical and the other of said surfaces being substantially flat, a universal joint having one part thereof fixed to the outer end of said rod and another part thereof fixed to said thrust member, said thrust member during operation of said machine causing said rod to move said piston into said cylinder to create pressure therein and the outer end of said rod to move radially in respect to said cylinder and thereby cause said contact surface to roll upon said bearing surface, said piston and said rod each having an axial passage therein through which liquid may flow from said cylinder to said joint, and a sealing member fitted in the passage

in said piston and urged by the pressure in said cylinder against the end of said rod to prevent any substantial amount of liquid from leaking out of said passages into said bore and having a duct extending therethrough and communicating with the passage in said rod.

2. A piston assembly according to claim 1 in which the passage in said rod is reduced to a very small diameter adjacent to said contact surface, said sealing member has the end thereof which engages said contact surface chamfered to reduce the area of said end, and said duct is very small in diameter adjacent to the end of said sealing member which engages said contact surface.

3. A piston assembly according to claim 1 including means for limiting the distance said sealing member can move away from said bearing surface during assembly of said machine and during any time that there is no pressure in said cylinder.

4. A piston assembly for a hydrodynamic machine having a cylinder and means including a thrust member for reciprocating a piston in said cylinder, said piston assembly comprising a piston fitted in said cylinder and having an axial bore extending into it from its outer end, a bearing surface on said piston at the inner end of said bore, a piston rod extending into said bore and having upon the inner end thereof a contact surface which engages said bearing surface, one of said surfaces being spherical and the other of said surfaces being substantially flat, a universal joint having one part thereof fixed to the outer end of said rod and another part thereof fixed to said thrust member, said thrust member during operation of said machine causing said rod to move said piston into said cylinder to create pressure therein and the outer end of said rod to move radially in respect to said cylinder and thereby cause said contact surface to roll upon said bearing surface, said piston and said rod each having an axial passage therein through which liquid may flow from said cylinder to said joint, a recessed seat formed in the end face of said rod around the passage therein, and a sealing member fitted in the passage in said piston and urged by the pressure in said cylinder against said seat to prevent any substantial amount of liquid from leaking out of said passages into said bore and having a duct extending therethrough and communicating with the passage in said rod, the end of said sealing member which engages said seat being closely fitted to said seat.

5. A piston assembly according to claim 4 including means for limiting the distance said sealing member can move away from said bearing surface during assembly of said machine and dur-

ing any time that there is no pressure in said cylinder.

6. A piston assembly for a hydrodynamic machine having a cylinder and means including a thrust member for reciprocating a piston in said cylinder, said piston assembly comprising a piston fitted in said cylinder and having an axial bore extending into it from its outer end, a substantially flat bearing surface on said piston at the inner end of said bore, a piston rod extending into said bore and having upon the inner end thereof a contact surface which engages said bearing surface, one of said surfaces being spherical and the other of said surfaces being substantially flat, a universal joint having one part thereof fixed to the outer end of said rod and another part thereof fixed to said thrust member, said thrust member during operation of said machine causing said rod to move said piston into said cylinder to create pressure therein and the outer end of said rod to move radially in respect to said cylinder and thereby cause said contact surface to roll upon said bearing surface, said piston and said rod each having an axial passage therein through which liquid may flow from said cylinder to said joint, a recessed seat formed in the end face of said rod around the passage therein, and a sealing member fitted in the passage in said piston and urged by the pressure in said cylinder against said seat to prevent any substantial amount of liquid from leaking out of said passages into said bore and having a duct extending therethrough and communicating with the passage in said rod, said sealing member including two parts the first of which is closely fitted in the passage in said piston and has a recessed seat on the end thereof toward said rod and the other of which is loosely fitted in the passage in said piston and has convex spherical seats arranged upon both of its ends and in contact with the recessed seats on said rod and on said first part.

7. A piston assembly according to claim 6 including means for limiting the distance said sealing member can move away from said bearing surface during assembly of said machine and during any time that there is no pressure in said cylinder.

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